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54 Power transmitting system for a four-wheel drive vehicle.

57 A torque distribution deciding control unit (50) a planetary gear device (20) as a central differential and an oil hydraulic clutch (27) having opposed disks are provided on a motor vehicle. The clutch (27) is disposed between output members of the planetary gear device, whereby torque is transmitted to front and rear wheels (10, 14) at a distribution ratio decided by the torque transmission of the clutch. The control unit operates the clutch so as to distribute the torque at a ratio in accordance with slipping of either of the front wheels so as to reduce the torque to the slipping wheels.

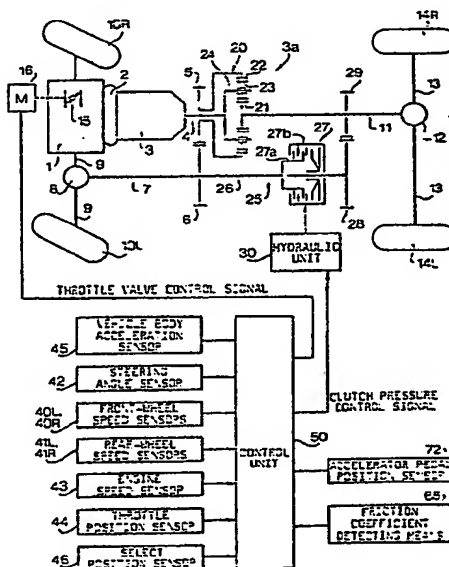


FIG. 1

Description

TRANSMISSION SYSTEM FOR A FOUR-WHEEL DRIVE VEHICLE

The present invention relates to a system for transmitting the power of an engine to four wheels of a four-wheel drive vehicle, and more particularly to a system for controlling traction of the vehicle.

Conventional fixed or full time four wheel drive vehicles have a differential acting between the front and rear wheels to distribute torque.

It is known that the distribution ratio of torque has influences on driveability, steerability, stability, starting characteristic and others. For example, the torque is split in accordance with the distribution of load on the front and rear wheels of the vehicle for good traction. If the torque distribution is controlled during cornering, a stable cornering characteristic can be obtained. Accordingly, the understeer characteristic encountered in a vehicle driven by front wheels and the oversteer characteristic encountered in a vehicle driven by rear wheels can be restrained. Further, on slippery roads, the distribution of the torque to the rear wheels can be increased so as to ensure safe driving without locking the differential mechanism.

Japanese Patent Laid Open No. 55-83617 discloses a transmitting system, in which the distribution ratio of torque to front and rear wheels is controlled electronically with a clutch or brake.

However, the system only has the central differential as a means for distributing the torque. Therefore, the distribution ratio is constant. In addition, the system can only select a two-wheel drive mode, four-wheel drive mode or a differential locking mode and is not intended for controlling the distribution ratio of torque.

However, although two-wheel slip can be prevented by controlling the torque distribution ratio, slip of all four wheels can not be prevented by the torque distribution control. In order to restrain the slipping of four wheels, for example, the power of the engine is reduced.

The object of the present invention is to provide a system to control the distribution ratio of torque to front wheels and rear wheels and engine power in accordance with various driving conditions of a four-wheel drive vehicle.

According to the present invention, there is provided a power transmitting system for a four-wheel drive vehicle comprising a differential for dividing the torque of an engine at a predetermined ratio, power trains for transmitting the torque to front and rear wheels of the vehicle, and a control device including a clutch arranged to act between the output members of the differential, for changing the distribution ratio, and a clutch actuator characterised by the provision of:

detecting means for detecting driving conditions of the vehicle including; speeds of the front and rear wheels and speed of the vehicle and for producing condition signals dependant on the detected conditions;

deciding means responsive to the condition signal for determining a desired torque distribution ratio

and for producing a distribution ratio signal representing the desired torque distribution ratio;
an antilock brake system producing an antilock brake signal;
two-wheel slip detecting means responsive to the condition signals for producing a two-wheel slip signal when either of the front wheels or rear wheels are slipping;
four-wheel slip detecting means responsive to the condition signals in the absence of the two-wheel slip signal for producing a four-wheel slip signal;
the clutch actuator responding to the two-wheel slip signal to engage the clutch so as to reduce the torque applied to the slipping wheels;
correcting means responsive to the antilock brake signal for correcting clutch torque; and
throttle actuator means responsive to the four-wheel slip signal to at least partly close the throttle valve of the engine.

The system has two-wheel slip detecting means responsive to the condition signals for producing a two-wheel slip signal when either of the front wheels or rear wheels are slipping, four-wheel slip detecting means responsive to the condition signals at nonexistence of the two-wheel slip signal for producing a four-wheel slip signal, first actuator means responsive to the two-wheel slip signal for engaging the clutch so as to reduce the split torque to the slipping wheels, and second actuator means responsive to the four-wheel slip signal for reducing opening degree of a throttle valve of the engine.

In an aspect of the invention, the clutch is an oil hydraulic clutch having opposite disks which operates to transmit torque by difference between speeds of opposite disks and by difference of pressure supplying to the clutch, and the central differential is a planetary gear.

The other objects and features of this invention will become understood from the following description with reference to the accompanying drawings.

BRIEF DESCRIPTION OF DRAWINGS

Fig. 1 is a schematic diagram showing a system according to the present invention;

Fig. 2a shows a hydraulic circuit for controlling a hydraulic clutch;

Fig. 2b is a graph showing a characteristic of a clutch pressure;

Figs. 3a and 3b show a block diagram of a control unit of an embodiment of the present invention;

Fig. 4a is a graph showing a torque distribution ratio in accordance with the vehicle speed and steering angle;

Fig. 4b is a graph showing engine torques in accordance with engine speed and throttle valve opening degree;

Fig. 4c is a graph showing a characteristic of torque ratio in accordance with speed ratio;

Fig. 4d is a graph showing a characteristic of

clutch torque in accordance with clutch pressure;

Fig. 4e is a graph showing desired speed differences between front and rear wheels in accordance with vehicle speed and steering angle in the embodiment of the present invention;

Fig. 4f is a graph showing a desired throttle valve opening degree in relation to an accelerator pedal depressing degree;

Fig. 4g is a graph showing a correcting quantity in relation to an acceleration of a vehicle body; and

Figs. 5a and 5b show a flowchart showing the operation of the control unit.

DETAILED DESCRIPTION OF THE PREFERRED EMBODIMENT

Referring to Fig. 1 showing a power transmission system for a four-wheel drive vehicle, reference numeral 1 designates an engine mounted on a front portion of the vehicle. The power of the engine 1 is transmitted to a transfer device 3a through a torque converter 2 and an automatic transmission 3. In the transfer device 3a, rotation of an output shaft 4 of the transmission 3 is transmitted to a central differential 20. The central differential 20 is a planetary gear device and comprises a sun gear 21, a ring gear 22 having internal teeth, a carrier 24, and planet pinions 23 rotatably supported on the carrier 24 and engaged with the both gears 21 and 22. The carrier 24 is connected with the output shaft 4. The ring gear 22 is connected to a gear 5 rotatably mounted on the shaft 4 and in mesh with a gear 6 having the same diameter as the gear 5 and fixedly mounted on a front drive shaft 7. The front drive shaft 7 is operatively connected to a front differential 8 so as to drive front wheels 10R and 10L through axles 9. On the other hand, the sun gear 21 is fixedly mounted on a rear drive shaft 11 so as to drive rear wheels 14R and 14L through a rear differential 12 and axles 13. Thus, the central differential 20 operates to split the torque of the engine at a ratio determined by the gear ratio thereof. The split torque is distributed to the front and rear wheels by the transfer device 3a as described hereinafter. The difference between speeds of front and rear wheels is absorbed by the central differential.

In the present arrangement of the transmission system, since static load on front part of the vehicle is larger than that of rear part, the central differential 20 is adapted to transmit a large torque to the front wheels through the ring gear 22 having a large diameter and to transmit a smaller torque to the rear wheels through the sun gear 21 having a small diameter.

In the system of the present invention, a torque split control device 25 is provided over the central differential 20. The torque split control device 25 includes gears 5 and 6, a bypass shaft 26 coaxially secured to the front drive shaft 7, a gear 29 secured to the rear drive shaft 11, and a gear 28 in mesh with the gear 29. The gears 28 and 29 are arranged to have a gear ratio which is slightly smaller than 1. The shaft 26 is connected to a hub 27a of an oil hydraulic

clutch 27 and gear 28 is secured to a drum 27b of the clutch 27. The clutch has a plurality of disks secured to the hub 27a and drum 27b, respectively.

The clutch 27 is arranged to transmit the torque from a higher speed disk side to a lower speed disk side. When pressurized oil is supplied to the clutch 27 from a hydraulic unit 30, a clutch torque is generated therein, thereby engaging the clutch 27. Since the drum 27b is rotated at a slightly smaller rate than the hub 27a, torque dependent on the clutch torque is transferred from the hub 27a to the drum 27b and to the rear wheels. Thus, torques T_F and T_R transmitted to the front and rear drive shafts 7 and 11 are respectively expressed as follows.

$$T_F = \gamma \cdot T_i - T_C \quad (1)$$

$$T_R = (1 - \gamma) \cdot T_i + K T_C \quad (2)$$

where T_i is input torque transmitted to the central differential 20, γ is distribution ratio for the front wheels predetermined by the central differential 20, K is the gear ratio of gears 28, 29, and T_C is the clutch torque.

Accordingly, as the clutch torque T_C increases, the distribution ratio for the front torque T_F becomes smaller than the distribution ratio γ . On the other hand, the distribution ratio of the rear torque T_R becomes larger than the ratio $(1 - \gamma)$. Thus, the torque distribution can be continuously varied.

On a throttle valve 15 of the engine 1, an actuator 16 such as a motor is attached for electrically actuating the throttle valve.

An electronic control system has left and right front-wheel speed sensors 40L and 40R, left and right rear-wheel speed sensors 41L and 41R, steering angle sensor 42, engine speed sensor 43, throttle position sensor 44, vehicle body acceleration sensor 45 and a select position sensor 46 for determining the selected gear in the automatic transmission 3, an accelerator pedal position sensor 72 and a friction coefficient detecting means 65. The vehicle body acceleration sensor 45 has a ground speed sensor and produces a vehicle body acceleration signal based on the ground speed.

A control unit 50 which is supplied with outputs of the sensors 40 to 46 and 72, and the friction coefficient detecting means 65 produces an output signal which is applied to the hydraulic unit 30 to control oil pressure applied to the oil clutch 27.

The hydraulic unit 30 for the clutch 27 is described hereinafter with reference to Fig. 2a. The hydraulic unit 30 comprises a pressure regulator valve 32, a transfer control valve 36 and a solenoid operated valve 34. The pressure regulator valve 32 is communicated with a passage 31 through which a line pressure of the automatic transmission 3 is supplied. The regulator valve 32 operates to provide a constant pilot pressure in a passage 33 by controlling the oil supplied to the passage 33 and drained therefrom. The passage 33 is communicated with the solenoid operated valve 34 and to an end of the transfer control valve 36 through a passage 35. The transfer control valve 36 is communicated with the passage 31 and with the clutch 27 through a passage 37. The solenoid operated valve 34 is operated by pulses from the control unit 50 at a duty ratio determined therein, thereby controlling the

drain of the oil to provide a control pressure P_c . The control pressure P_c is applied to an end of a spool 36a of the transfer control valve 36 to control the oil supplied to the clutch 27 so as to control the clutch pressure (torque).

Fig. 2b shows the relationship between the clutch pressure P_T , control pressure P_c and duty ratio. When the duty ratio is at 0%, the solenoid operated valve 34 does not drain the oil. Accordingly the control pressure P_c is at the highest value which is equal to the pilot pressure. As the duty ratio increases, the control pressure P_c reduces as shown by a broken line. On the other hand, the clutch pressure P_T is zero in a small duty ratio range and increases with increase of the duty ratio.

Referring to Figs. 3a and 3b, the control unit 50 has a front wheel speed calculator 51 to which speeds N_{FL} and N_{FR} from the left and right front wheel sensors 40L and 40R are applied, and a rear wheel speed calculator 52 to which speeds N_{RL} and N_{RR} from the left and right rear wheel speed sensors 41L and 41R are applied. Front wheel speed N_F and rear wheel speed N_R are obtained from the following equations.

$$N_F = (N_{FL} + N_{FR})/2$$

$$N_R = (N_{RL} + N_{RR})/2$$

The speeds N_F and N_R are fed to a vehicle wheel speed calculator 53 where the average speed N of the four wheels is calculated as follows.

$$N = (N_F + N_R)/2$$

The speed N is applied to a vehicle speed calculator 54 where vehicle speed V is obtained from the speed N and diameter of tires of the vehicle.

The vehicle speed V and the steering angle λ detected by the steering angle sensor 42 are applied to a desired torque distribution ratio table 55. The desired torque distribution ratio table 55 is stored with a plurality of torque distribution ratios S_R for the rear wheels provided from a graph shown in Fig. 4a. The table provides a distribution ratio S_R in accordance with the steering angle λ and the vehicle speed V . When the vehicle is steered at a small radius in a middle speed range, the desired torque distribution ratio S_R is increased so as to improve the steerability of the vehicle. To the contrary, if the vehicle is steered at a large radius at a high speed, the desired distribution ratio S_R becomes small in order to ensure stability of the vehicle. In other regions, the distribution ratio is set so as to comply both with the steerability and stability. The desired distribution ratio S_R is applied to an acceleration correcting section 56 to which an acceleration G of the vehicle body from the acceleration sensor 45 is applied to correct the distribution ratio in accordance with the acceleration. If the acceleration G is zero, the vehicle is in a steady state so that no correction is performed.

Engine speed N_e and throttle opening degree θ from the engine speed sensor 43 and the throttle position sensor 44 are applied to an engine torque table 57 from which engine torque T_e is derived in accordance with N_e and θ as shown in a graph of Fig. 4b. The engine speed N_e , the vehicle wheel speed N and transmission ratio Gr detected by the select position sensor 46 are applied to a torque

converter speed ratio calculator 58. The calculator 58 calculates a turbine speed N_t , which is an output speed of the torque converter 2, from the vehicle wheel speed N and the transmission ratio Gr . Speed ratio R_w between the engine speed N_e which is an input speed of the torque converter 2 and the output speed N_t is

$$R_w = N_t/N_e.$$

The speed ratio R_w is applied to a torque converter torque ratio calculator 59 from which a torque ratio R_t of the torque converter is derived as shown in Fig. 4c. The torque ratio R_t is applied to a central differential input torque calculator 60 to which engine torque T_e and transmission ratio Gr are also applied. The input torque T_i to the central differential 20 is calculated as follows.

$$T_i = T_e \cdot R_t \cdot Gr$$

The desired torque distribution ratio S_R and the torque T_i are applied to a clutch pressure calculator 61. The distribution ratio S_R can be expressed as below.

$$S_R = T_R/(T_F + T_R) \quad (3)$$

In accordance with the afore-described equations (1), (2) and (3), the clutch torque T_c can be expressed as

$$T_c = f(S_R, T_i)$$

Accordingly, the clutch torque increased with the increase of the torque distribution ratio S_R and input torque T_i . The clutch pressure P_T is an increasing function of the clutch torque T_c , depending on the number of disks in the clutch 27 and friction coefficient between the disks. Thus, the clutch pressure P_T is expressed as $P_T = g(T_c)$ as shown in a graph of Fig. 4d. The clutch pressure P_T is applied to a duty ratio setting section 62 where a duty ratio signal corresponding to the clutch pressure P_T applied to the hydraulic clutch 27 is obtained.

The control unit 50 further has a desired speed difference providing section 63, an actual speed difference calculator 64 and a two-wheel slip determining section 66. The desired speed difference providing section 63 provides a desired speed difference ΔN_d between the front and rear wheels in dependence on the vehicle speed V and steering angle λ . As shown in a graph of Fig. 4e, the desired speed difference ΔN_d increases with the increase of the steering angle λ . The difference ΔN_d also gradually increases with the increase of the vehicle speed V in a low vehicle speed region, and gradually decreases with the further increase of the vehicle speed beyond a predetermined vehicle speed. When the steering angle is zero for driving the vehicle straight-ahead, the desired speed difference ΔN_d is set at zero, so as to render the vehicle follow an optimum path without slipping. Additionally, such an optimum path varies in dependence on conditions of the road surface and engine torque. Thus, in the present embodiment, the friction coefficient μ is applied to the desired speed difference providing section 63 to correct the speed difference ΔN_d , that is by decreasing the difference as the friction coefficient decreases.

The actual speed difference calculator 64 calculates an actual difference ΔN of the front and rear

wheel speeds as follows.

$$\Delta N = N_F - N_R$$

The desired and actual speed differences ΔN_d and ΔN are applied to the two-wheel slip determining section 66 where it is determined that either of front wheels or rear wheels are slipping when the actual difference ΔN deviates from the desired difference. Namely when the actual speed difference ΔN is under the following conditions.

$$\Delta N > \Delta N_d + K1$$

$$\Delta N < \Delta N_d - K2$$

where $K1$ and $K2$ are constants corresponding to a dead zone for determining the slipping which is decided by the accuracy of the speed sensors, and other. The two-wheel slip signal from the slip determining section 66 is applied to a correcting section 67 to correct the clutch pressure in accordance with the slipping rate.

A system for controlling the traction of the engine is described hereinafter. An accelerator pedal depressing degree ϕ detected by the accelerator pedal position sensor 72 is applied to a desired throttle opening degree table 73 where a plurality of desired throttle opening degrees θ_d , which increase with the accelerator pedal depressing degree ϕ as shown in Fig. 4f, are stored. The desired throttle opening degree θ_d is applied through a correcting section 75 to a throttle valve actuator operating quantity calculator 74, where the actual throttle opening degree from the sensor 44 is applied. The calculator 74 produces the throttle valve control signal in dependence on a deviation of the actual throttle opening degree θ from the desired opening degree θ_d .

The desired throttle opening degree θ_d is corrected in the correcting section 75 in accordance with a signal from a four-wheel slip/grip deciding section 71, to which the output signal of the two-wheel slip deciding section 66, the vehicle wheel speed N and the acceleration G of the vehicle body are fed. The deciding section 71 determines that the four wheel of the vehicle are slipping or that the four wheels grip the road surface when the actual speed difference ΔN between front and rear wheels is in a range of $\Delta N_d - K2$ and $\Delta N_d + K1$ ($\Delta N_d - K2 \leq \Delta N \leq \Delta N_d + K1$), which means that the two-wheel slip signal does not exist. A difference ΔG between the acceleration G of the vehicle body and an acceleration G_N of the vehicle wheels, which is a differential of the vehicle wheel speed N , is calculated ($\Delta G = G_N - G$). When the difference ΔG is larger than a predetermined slip reference value ΔG_S ($\Delta G \geq \Delta G_S$), it is determined that the four wheels of the vehicle are slipping ($\Delta G_S > 0$). On the other hand, when the difference ΔG is smaller than a predetermined grip reference value $\Delta G_S'$ ($\Delta G \leq \Delta G_S'$), it is determined that the four wheels resumed gripping of the road surface after the slipping ($\Delta G_S' < 0$). The four-wheel slip and grip may be determined in dependence on a difference ΔN between the vehicle wheel speed N and a vehicle body speed N_G which is an integral of the acceleration G of vehicle body, predetermined slip reference value ΔN_S and predetermined grip reference value $\Delta N_S'$ in the same manner as the above described method.

When a slip signal is applied to the correcting section 75, the desired throttle opening degree θ_d is decreased, and when a grip signal is applied after the slipping, the original desired throttle opening degree θ_d is restored.

The correcting quantity is set in relation to the acceleration G of the vehicle body as shown in a graph of Fig. 4g. If the four-wheel slip occurs when the acceleration G is small, assuming that slipping rate is large on account of the low friction coefficient of the road surface, the throttle opening degree is largely decreased to reduce the traction.

When the wheels skid while the brake is actuated, an antilock brake system (ABS) operates to reduce a braking effect, in response to output signals of front and rear wheels speed calculators 51 and 52 and vehicle speed calculator 54. In the ABS, since the braking effect is influenced by the engine torque, controlling of the transmitted torque, that is the traction control, is not preferable. Therefore, an antilock brake signal from the ABS 77 is applied to the correction section 67 so as to render the clutch pressure P_C in the clutch 27 zero. Accordingly, the torque is split in accordance with the basic torque distribution ratio dependent on the central differential 20.

When the drive range is selected, the output torque of the engine is transmitted through the torque converter 2 and the transmission 3 to the central differential 20 at which the torque is divided to the ring gear 22 and sun gear 21, for example at a ratio 60:40, in accordance with a static load of the vehicle. The divided torques are transmitted to front and rear drive shafts 7 and 11. At the same time, the torques are transmitted to hub 27a of the oil clutch 27 through gears 5, 6 and shaft 26 and to the drum 27b through gears 29 and 28, respectively. Since the speed of the drum 27b is reduced by the gear ratio K between gears 29 and 28, the torque is transmitted from hub 27a of higher speed side to the drum 27b of lower speed side.

During a steady driving state where the vehicle is not accelerated, that is, when the acceleration G is substantially zero, the desired torque distribution ratio S_R for the rear wheels is held at a basic ratio of 40%. Therefore, the clutch pressure P_r calculated at the calculator 61 is zero. Accordingly, duty ratio of the pulses applied to the solenoid operated valve 34 is set at zero at the duty ratio setting section 62. The control pressure P_C is at maximum, thereby draining the oil in the clutch 27. Therefore, clutch pressure P_r in the clutch 27 becomes zero so that the clutch is disengaged so as to render the members of the torque split control device 25 idle. Accordingly, the input torque is split to front and rear wheels 10R, 10L, 14R, 14L in accordance with the basic ratio.

If the vehicle is steered at a relatively small radius by a large steering angle while driven at a steady state at middle speed, the desired torque distribution ratio S_R is increased. The control unit 50 applies corresponding pulse having a larger duty ratio, thereby draining the oil from the solenoid valve 34. Accordingly, the control pressure P_C decreases, supplying a large amount of oil to the clutch 27. Consequently, a large clutch torque T_C is produced

in the clutch, so that a torque dependent on the clutch torque T_C is transmitted from hub 27a to drum 27b. Accordingly, the torque T_F to the front wheels is reduced by a torque based on the large clutch torque T_C . To the contrary, the torque T_R to the rear wheels is greatly increased by a torque based on the clutch torque and the gear ratio K at gears 28 and 29.

On the other hand, if the vehicle speed V is low and the steering angle λ is small, the transfer torque is decreased to reduce the torque to rear wheels.

When the vehicle is accelerated, the load on the vehicle moves to the rear part of the vehicle. In accordance with the movement of the load, the torque distribution ratio S_R from the table 55 is increased in the correcting section 56, so as to increase the rear torque T_R , even if the vehicle is driven straight. Therefore, traction is improved. If the input torque T_i becomes small, the clutch pressure P_T becomes small regardless of the value of the distribution ratio S_R so that tight corner braking can be prevented.

The above described control patterns are summarized as set out in the following table.

TABLE

	STRAIGHT-AHEAD DRIVING	CORNERING
STEADY STATE	BASIC DISTRIBUTION	DISTRIBUTION RATIO TO REAR WHEELS INCREASES
ACCELERATION	DISTRIBUTION RATIO TO REAR WHEELS INCREASES	

The system for preventing the two-wheel and four-wheel slips is described hereinafter with reference to a flowchart shown in Figs. 5a and 5b.

At a step S101, the steering angle λ and the vehicle speed V are read out. The front-wheel speed N_F and rear wheel speed N_R are calculated at steps S102 and S103, respectively. The desired speed difference ΔN_d is derived from the table 63 dependent on the steering angle λ and vehicle speed V at a step S104. At a step S105, it is determined whether the speed difference ΔN between the front wheels and rear wheels is larger than $\Delta N_d + K1$. If the difference ΔN is larger, meaning that only the front wheels are slipping, the program proceeds to a step S106, where clutch pressure P_T applied to the clutch 27 is increased so as to increase the torque distribution ratio for rear wheels. Accordingly, the differential operation of the central differential 20 is limited by the clutch 27. When the difference ΔN is smaller than $\Delta N_d + K1$, the program proceeds to a step S107 where it is determined whether the difference ΔN is smaller than $\Delta N_d - K2$. When the difference ΔN is smaller, the rear wheels are slipping so that it can be assumed that the clutch 27 generates a large clutch torque, thereby limiting the differential operation of

the central differential 20. Therefore, the clutch pressure P_T is decreased to increase the torque to front wheels at a step S108. Thus, the actual difference ΔN is converged to the desired difference ΔN_d by the feedback control. Since the differential is not locked, the skidding is prevented while keeping steerability.

When the front wheels or the rear wheels slip while the vehicle is driven straight ahead, the actual difference ΔN always becomes much larger than the desired difference ($|\Delta N| \gg |\Delta N_d|$). In the present embodiment, the vehicle can be got out the stuck state or driven by increasing the clutch pressure to limit the differential operation. Furthermore, on slippery roads, the difference ΔN_d is decreased dependent on the friction coefficient μ , so that the operation of the central differential 20 is limited, thereby preventing slipping.

When the difference is between $\Delta N_d + K1$ and $\Delta N_d - K2$ ($\Delta N_d - K2 \leq \Delta N \leq \Delta N_d + K1$), the traction control for preventing four-wheel slip is carried out. At a step S109, vehicle wheel speed N is calculated in dependence on the front wheel speed N_F and the rear wheel speed N_R , and at a step S110, vehicle wheel acceleration G_N for example, is calculated. The difference ΔG between the wheel acceleration G_N and vehicle body acceleration G is calculated at a step S111. At a step S112, it is determined whether flag indicating the four-wheel slip was set at the last program. If the flag is not set, the difference ΔG is compared with the predetermined slip value ΔG_s at a step S113. When the difference ΔG is smaller than the value ΔG_s , determining that the vehicle is steadily driven, the program goes to a step S114, where it is again determined if the flag is set or not. Since the flag is not set, the program proceeds to a step S115 where the desired throttle valve opening degree θ_d derived from the table 73 is fed to the actuator operating quantity calculator 74 without any correction. Accordingly, a control signal representing the operating quantity dependent on the desired degree θ_d and actual degree θ is applied to the actuator 16 of the throttle valve 15 so that the throttle valve is adjusted to the desired opening degree θ_d .

When the acceleration G_N of the vehicle wheels is much larger than the acceleration G of the vehicle body so that the difference ΔG is larger than the slip value ΔG_s , it is determined that the four wheels of the vehicle are slipping. Accordingly, the flag is set at a step S116. Determining that the flag is set at the step S114, the program goes to a step S117, where the desired opening degree θ_d is reduced in the correcting section 75. The actuator operating quantity is calculated in dependence on the corrected desired opening degree θ_d' and the corresponding signal is applied to the actuator 16. Consequently, the opening degree of the throttle valve 15 is decreased so as to decrease the torque of the engine 1, thereby reducing the vehicle wheel speed N and vehicle wheel acceleration G_N .

When the flag is detected at the step S112, the program goes to a step S118. Since the vehicle wheel acceleration G_N is reduced at the previous program, the difference ΔG between the vehicle

wheel acceleration G_N and the vehicle body acceleration G is reduced. The difference ΔG is now compared with the predetermined grip value $\Delta G_s'$. When the difference is larger, the four wheels are slipping so that the program proceeds to the step S117 through the step S114, repeating the afore-described operation.

When the difference ΔG is smaller than the grip value $\Delta G_s'$, it is determined that the four wheels grip the road surface. The flag is reset at a step S119. Accordingly, the program goes to the step S115 through the step S114, thereby terminating the correction in the correcting section 75. Thus, the throttle valve is controlled at the desired degree θ_d .

When the ABS is operated at the actuating of the brake, the antilock brake signal is applied to the correcting section 67 so that the clutch pressure P_r in the clutch 27 becomes zero. Therefore, the basic distribution ratio is resumed so as to effectively brake the vehicle by ABS.

The present invention may be modified to reduce the traction by braking the engine or controlling an ignition timing, thereby decreasing the engine torque.

In accordance with the present invention, the four-wheel slip or spin, which is a state of emergency where the slipping cannot be prevented by controlling the torque split to front and rear wheels, can be accurately determined in distinction from the two-wheel slip. Accordingly, the traction of the vehicle is decreased only at the four-wheel slip. Thus, each of the four wheels is equally and effectively prevented from slipping. In addition, the control of the torque distribution ratio is cancelled when the ABS is operated so as to reliably control the system.

While the presently preferred embodiment of the present invention has been shown and described, it is to be understood that this disclosure is for the purpose of illustration and that various changes and modifications may be made without departing from the scope of the invention as set forth in the appended claims.

Claims

1. A power transmitting system for a four-wheel drive vehicle comprising a differential (20) for dividing the torque of an engine at a predetermined ratio, power trains for transmitting the torque to front and rear wheels of the vehicle, and a control device including a clutch (27) arranged to act between the output members of the differential (20), for changing the distribution ratio, and a clutch actuator (30) characterised by the provision of:
 detecting means for detecting driving conditions of the vehicle including; speeds of the front and rear wheels and speed of the vehicle and for producing condition signals dependant on the detected conditions;
 deciding means (50) responsive to the condi-

tion signal for determining a desired torque distribution ratio and for producing a distribution ratio signal representing the desired torque distribution ratio;

an antilock brake system (77) producing an antilock brake signal;

two-wheel slip detecting means (66) responsive to the condition signals for producing a two-wheel slip signal when either of the front wheels or rear wheels are slipping;

four-wheel slip detecting means (71) responsive to the condition signals in the absence of the two-wheel slip signal for producing a four-wheel slip signal;

the clutch actuator (30) responding to the two-wheel slip signal to engage the clutch (27) so as to reduce the torque applied to the slipping wheels;

correcting means responsive to the antilock brake signal for correcting clutch torque; and throttle actuator means (16) responsive to the four-wheel slip signal to at least partly close the throttle valve (15) of the engine.

2. A system according to claim 1 characterised in that the clutch is an oil hydraulic clutch having opposed disks which operate to transmit torque according to a difference in the speeds of the disks and the actuating pressure applied to the clutch.

3. A system according to claim 1 or claim 2 characterised in that the differential is a planetary gear.

4. A system according to any preceding claim characterised in that the four-wheel slip signal is generated in dependance on the difference between vehicle body acceleration and acceleration of the vehicle wheels.

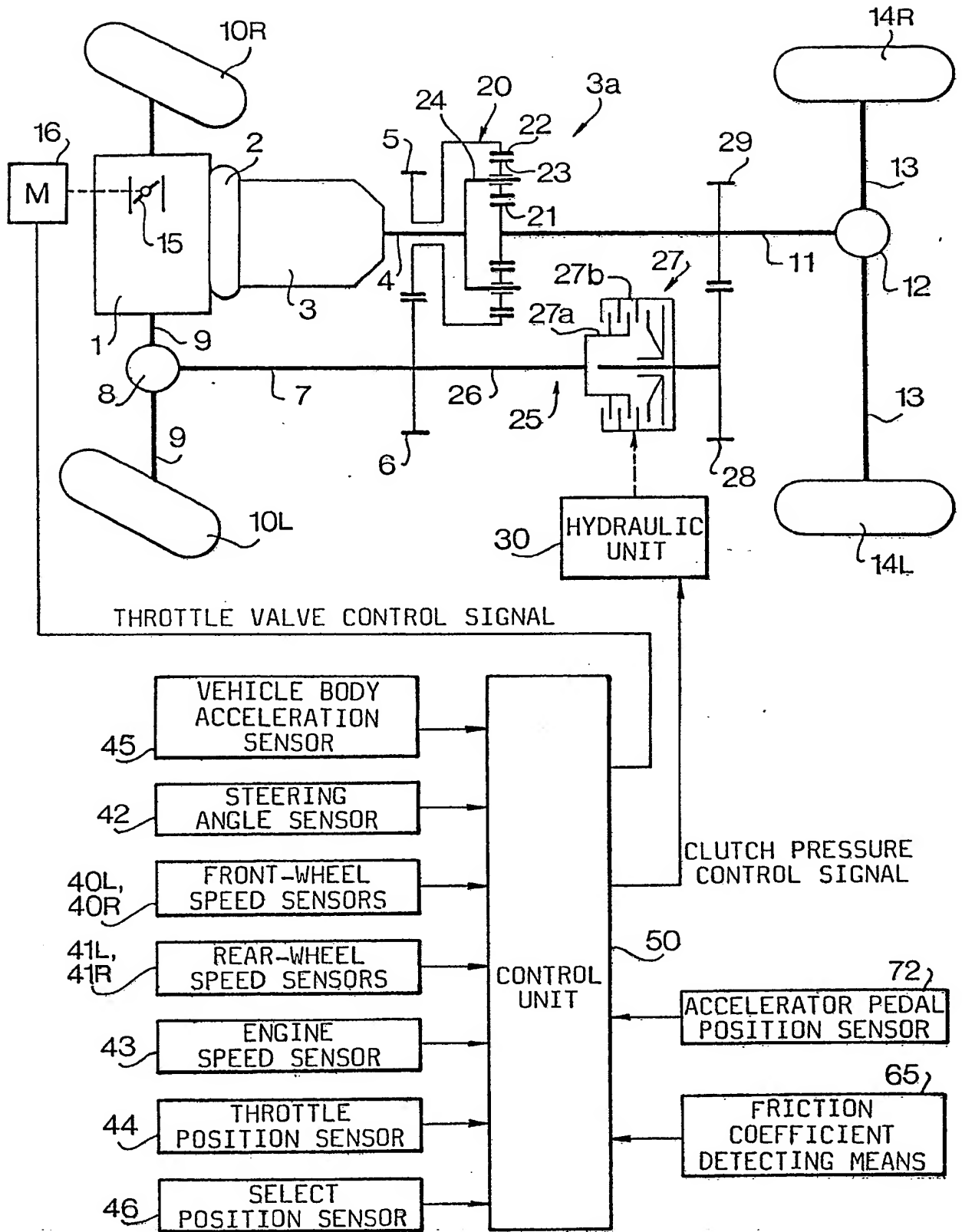


FIG. 1

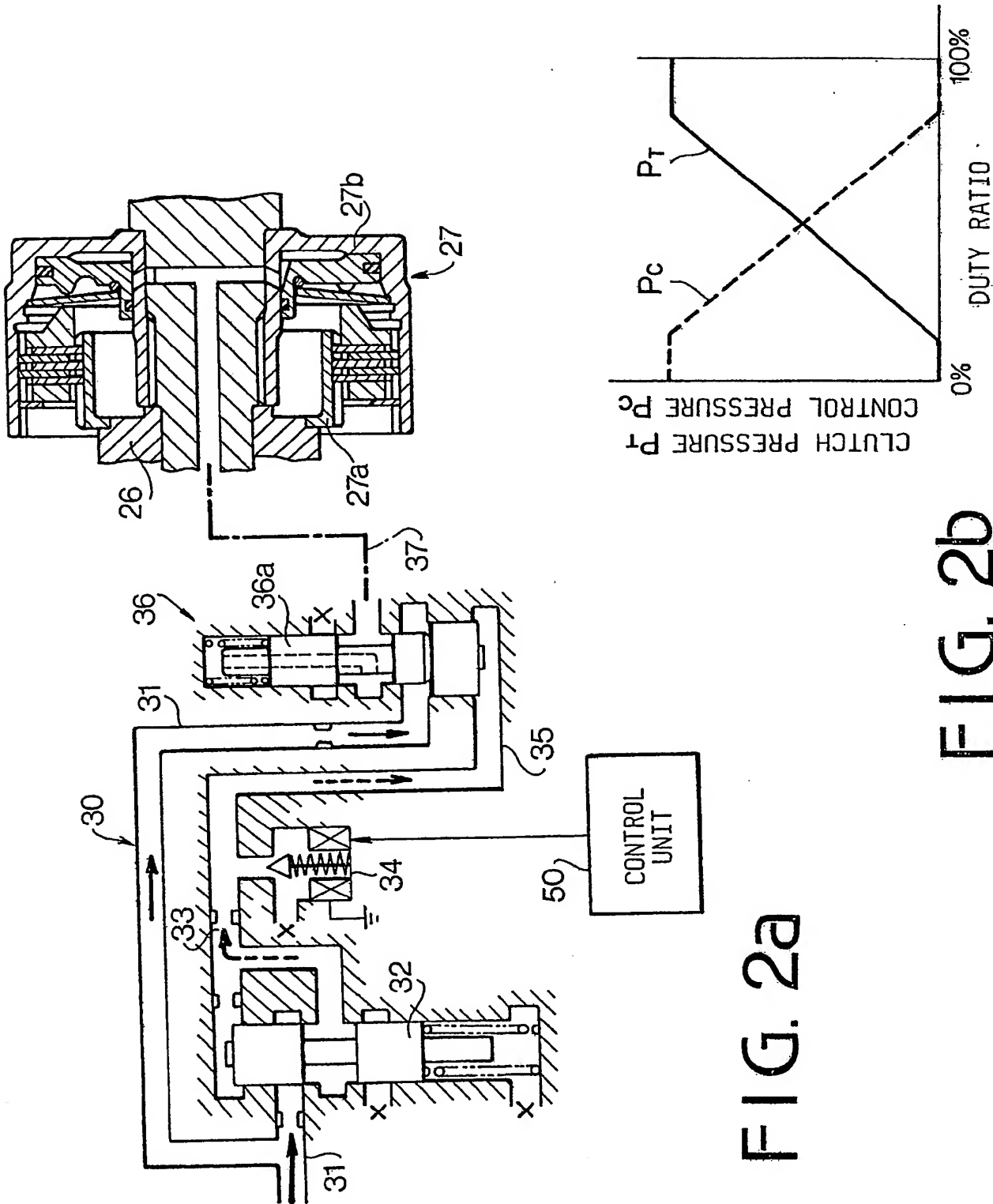


FIG. 2a

FIG. 2b

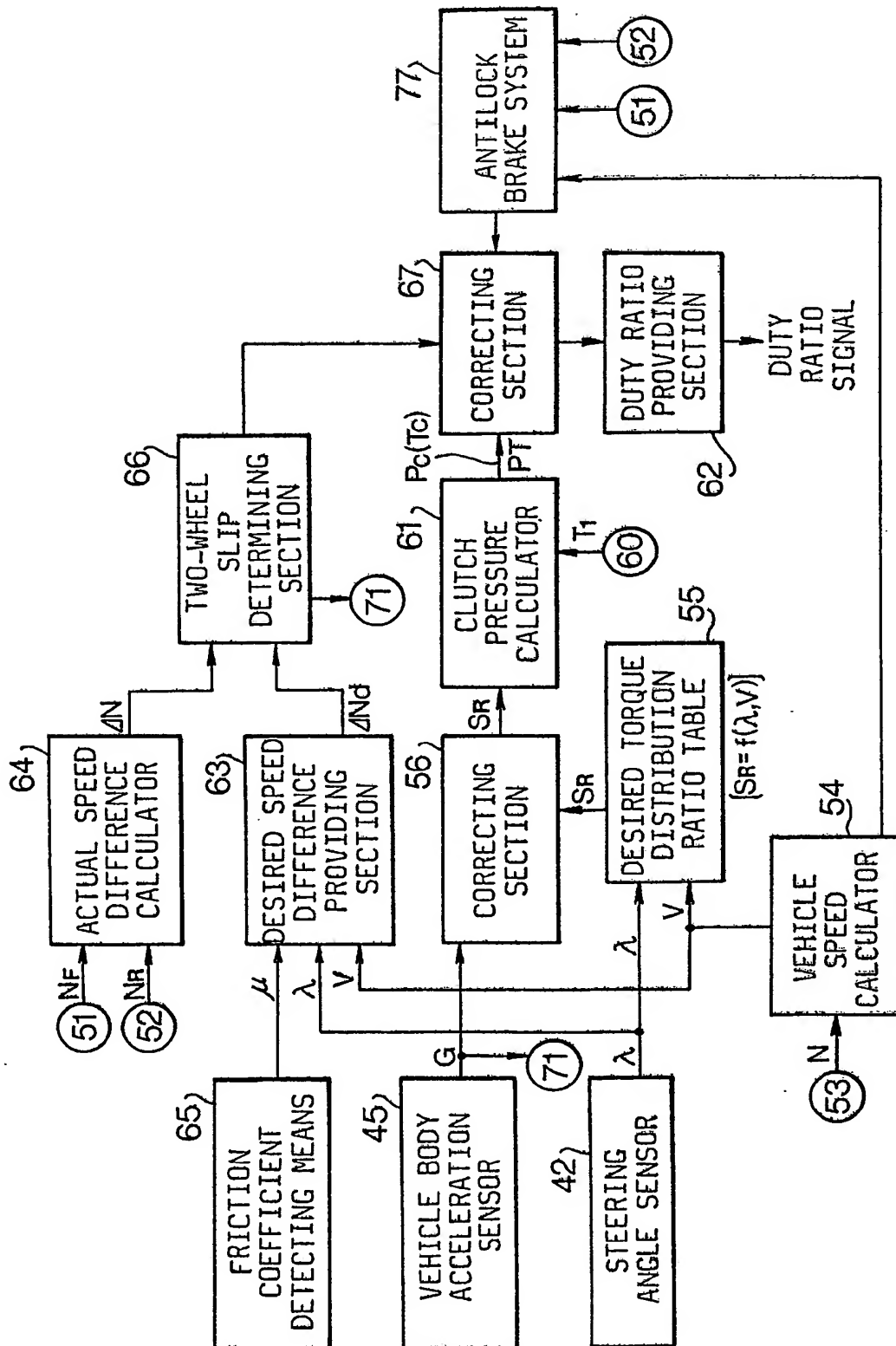


FIG. 3a

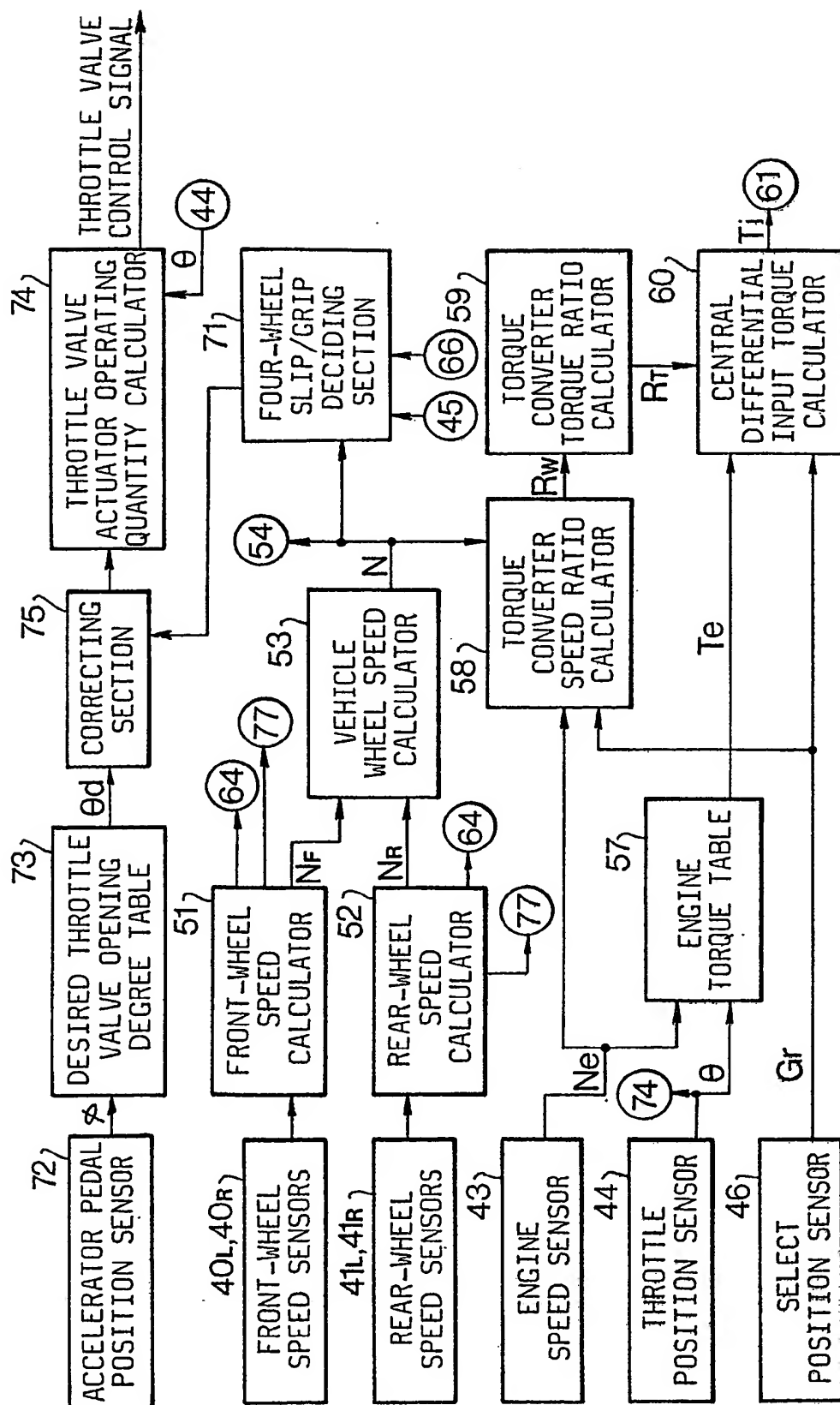


FIG. 3b

FIG. 4a

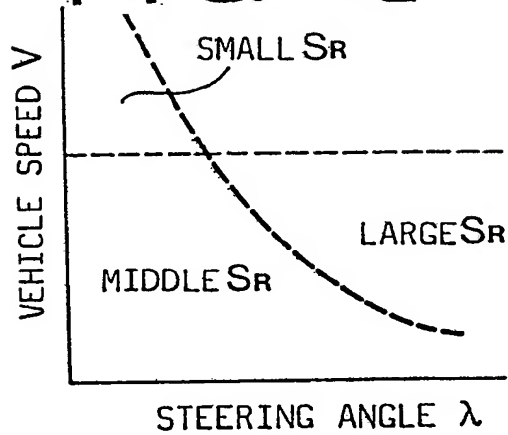


FIG. 4b

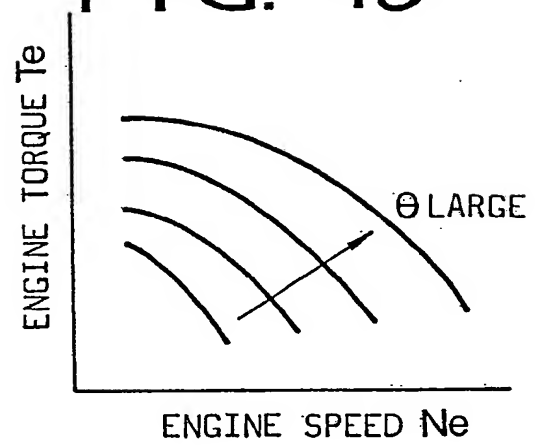


FIG. 4c

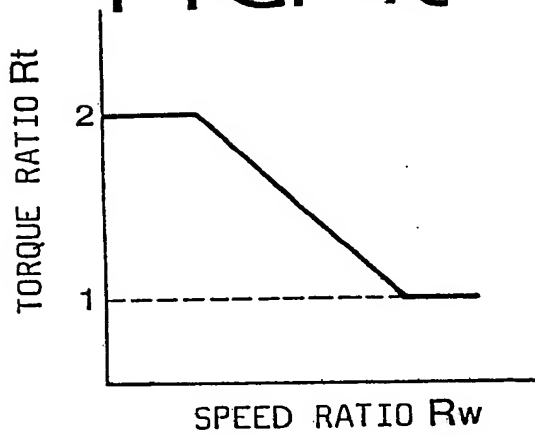
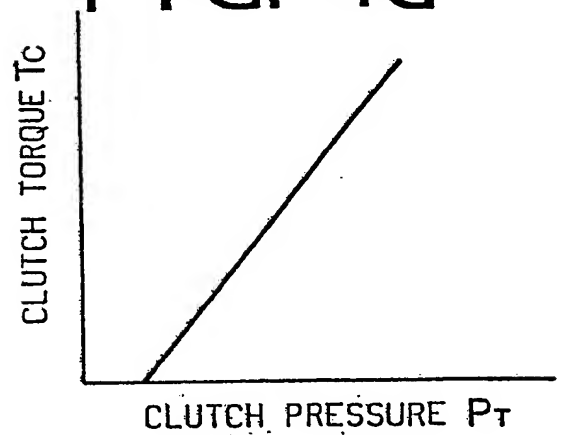
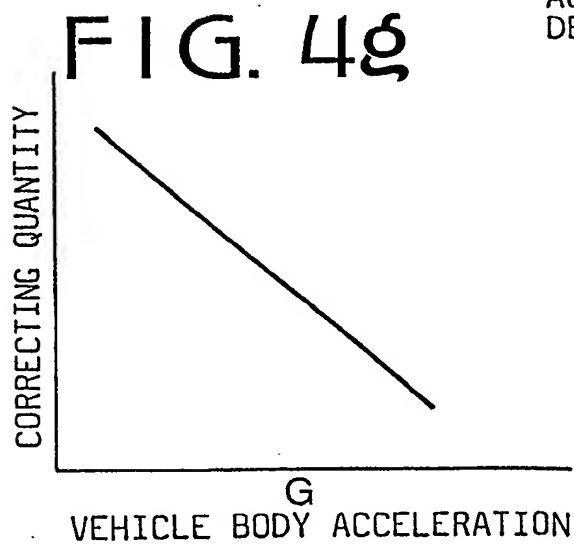
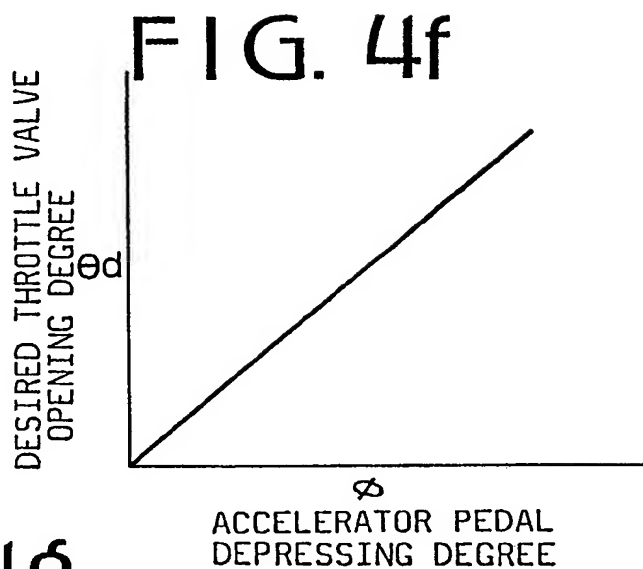
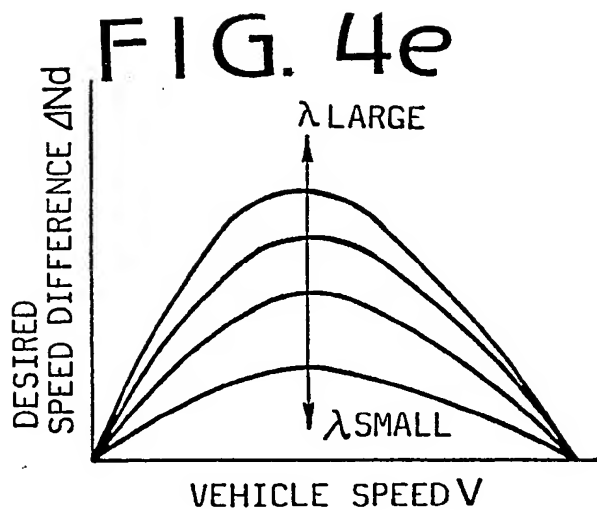


FIG. 4d





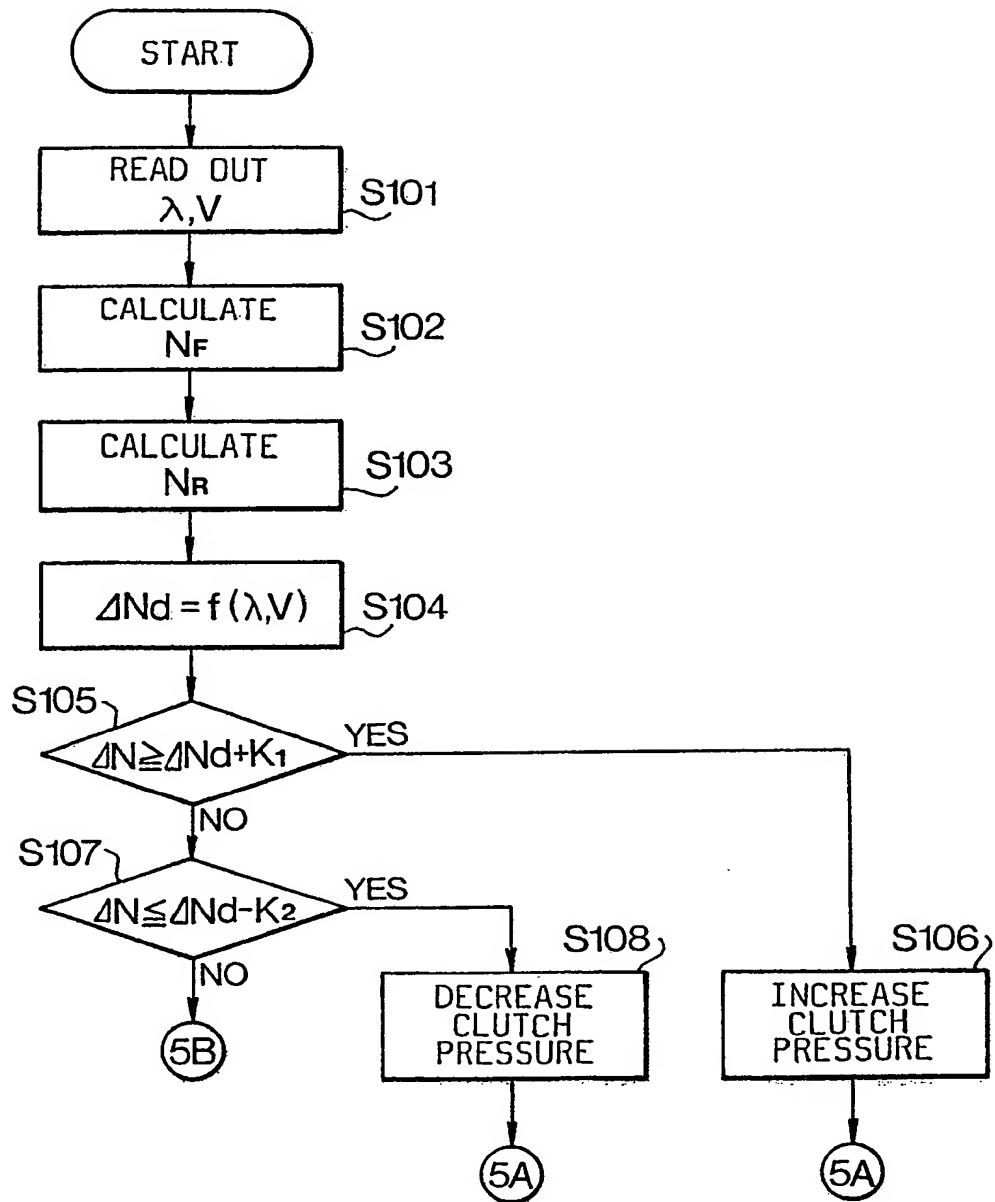


FIG. 5a

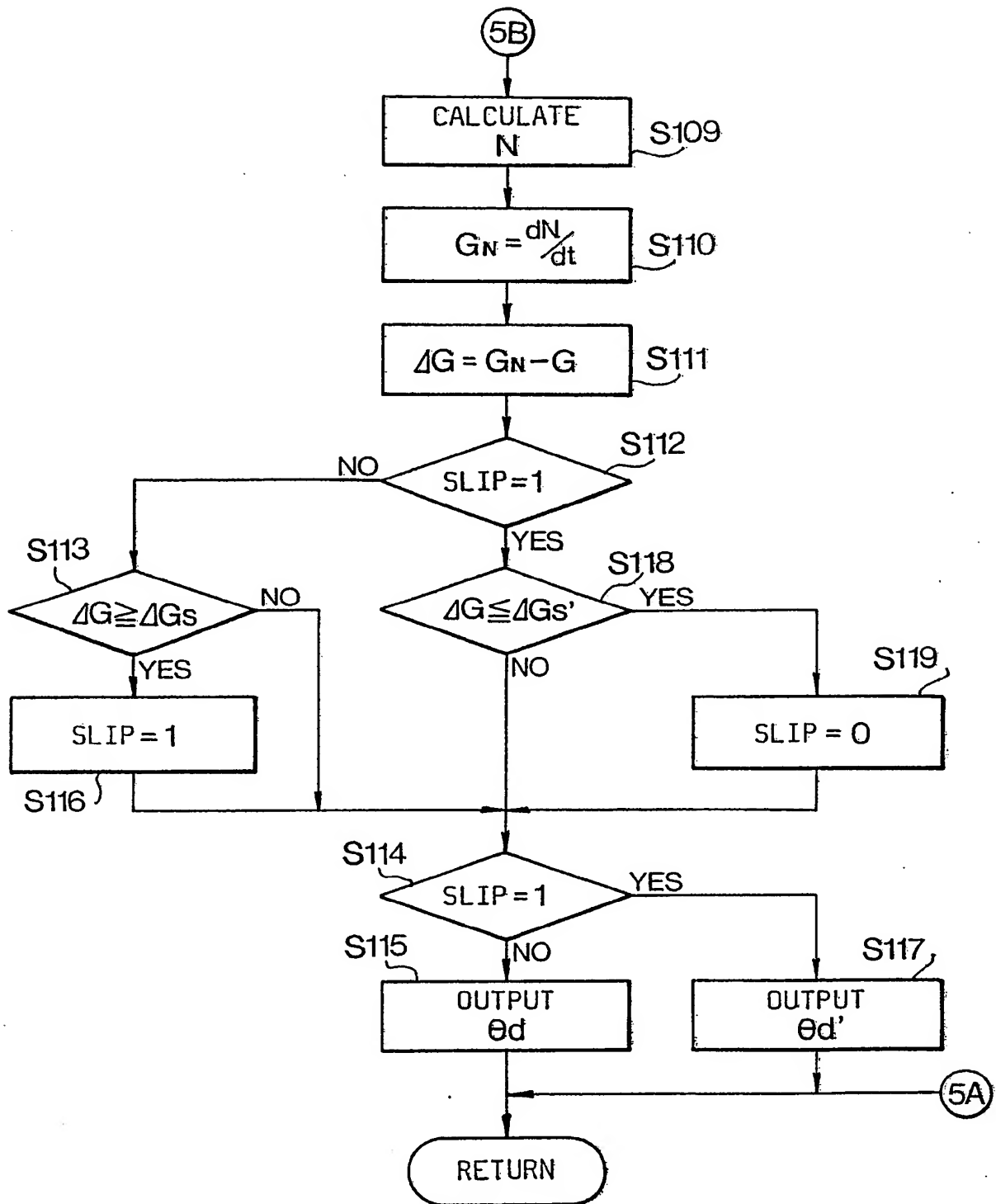


FIG. 5b

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54 **Power transmitting system for a four-wheel drive vehicle.**

57 A torque distribution deciding control unit (50) a planetary gear device (20) as a central differential and an oil hydraulic clutch (27) having opposed disks are provided on a motor vehicle. The clutch (27) is disposed between output members of the planetary gear device, whereby torque is transmitted to front and rear wheels (10, 14) at a distribution ration decided by the torque transmission of the clutch. The control unit operates the clutch so as to distribute the torque at a ratio in accordance with slipping of either of the front wheels so as to reduce the torque to the slipping wheels.

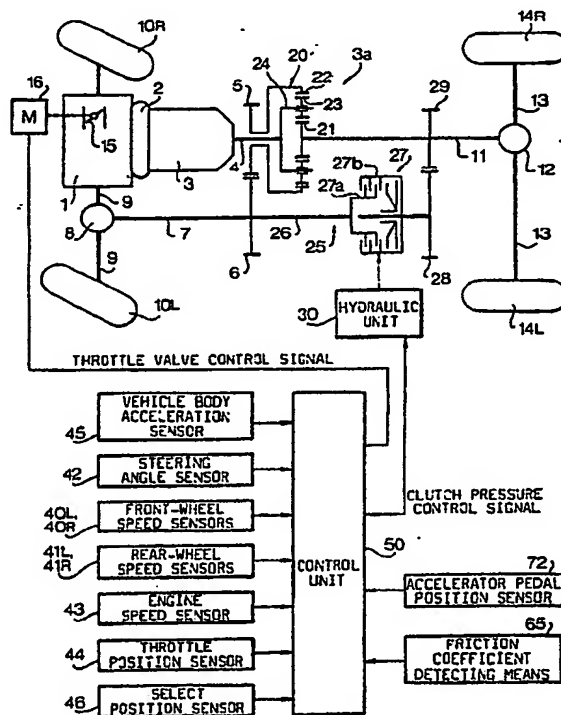


FIG. 1

EP 0 314 452 A3



European Patent
Office

EUROPEAN SEARCH REPORT

Application Number

EP 88 31 0056

DOCUMENTS CONSIDERED TO BE RELEVANT			
Category	Citation of document with indication, where appropriate, of relevant passages	Relevant to claim	CLASSIFICATION OF THE APPLICATION (Int. Cl.4)
X	DE-A-3427725 (VOLKSWAGEN) * page 8, paragraph 3 * * page 10, line 3 - page 11, line 15 * ---	1, 4	B60K28/16 B60K17/346
P,X	DE-A-3636260 (OPEL) * the whole document * ---	1	
A	DE-A-3626025 (NISSAN) * column 7, lines 22 - 62 * * column 11, lines 54 - 64; figures 1, 2, 3, 11 * ---	1, 2	
A	FR-A-2591958 (DAIMLER-BENZ) * the whole document * ---	1, 2	
A	PATENT ABSTRACTS OF JAPAN vol. 011, no. 295 (M-626) 24 September 87, & JP-A-62 085146 (NISSAN) 18 April 87, * the whole document * ---	1	
A	DE-A-3545717 (DAIMLER-BENZ) * the whole document * ---	1	TECHNICAL FIELDS SEARCHED (Int. Cl.4)
A	EP-A-0216318 (OPEL) * abstract; figure 3 * ---	1	B60K
A	DE-A-3706075 (FUJI JUKOGYO) -----		
The present search report has been drawn up for all claims			
Place of search BERLIN		Date of completion of the search 27 APRIL 1990	Examiner KRIEGER, P
CATEGORY OF CITED DOCUMENTS X : particularly relevant if taken alone Y : particularly relevant if combined with another document of the same category A : technological background O : non-written disclosure P : intermediate document T : theory or principle underlying the invention E : earlier patent document, but published on, or after the filing date D : document cited in the application L : document cited for other reasons & : member of the same patent family, corresponding document			

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